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Studying the Advisability of Using Gas-Turbine Unit Waste Gases for Heating Feed Water in a Steam Turbine Installation with a Type T-110/120-12.8 Turbine

A. D. Trukhnii*, G. D. Barinberg, and Yu. A. Rusetskii*****

**Moscow Power Institute (MEI), ul. Krasnokazarmennaya 14, Moscow, 111250 Russia*
***ZAO Urals Turbine Works (UTZ), ul. Frontovykh Brigad 18, Yekaterinburg, 620040 Russia*
****AO Teploenergoservis EK, P.O. box 522, Yekaterinburg, 620057 Russia*

Abstract—Results of calculation studying of a possibility of topping of a steam-turbine unit (STU) with a type T-110/120-12.8 turbine of the Urals Turbine Works (UTZ) by a gas-turbine unit (GTU) of 25-MW capacity the waste-gases heat of which is used to substitute for high-pressure bleedoffs of STU is considered. It is shown that this makes it possible to increase electric power up to 130 MW and reduce fuel consumption by 2.5–4.0% while operating in the condensing mode and by 1.5–2.0% in the cogenerating mode.

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One of the ways to increase the efficiency of steam power units (STUs) in operation is the use of heat of gas-turbine unit (GTU) waste gases for heating feed water with simultaneous delivery of the steam of the switched-off bleedoffs to the turbine flow path. For such a type of combined-cycle installation (CCI), the term *CCI with substituted regeneration* (CCI–R) is most widely used. Fuel savings in CCI–R relative to STU indexes are considerably smaller than those in utilization- or discharge-type CCIs. This is, first of all, due to small the "binary factor" of the combined cycle obtained. Using CCI–Rs, however, can be advisable from economic sense because of the following advantages of these installations:

(i) the short time to put them into operation,

(ii) small capital investments in the GTU (due to its relatively small power) and feed-water gas heater (FWGH),

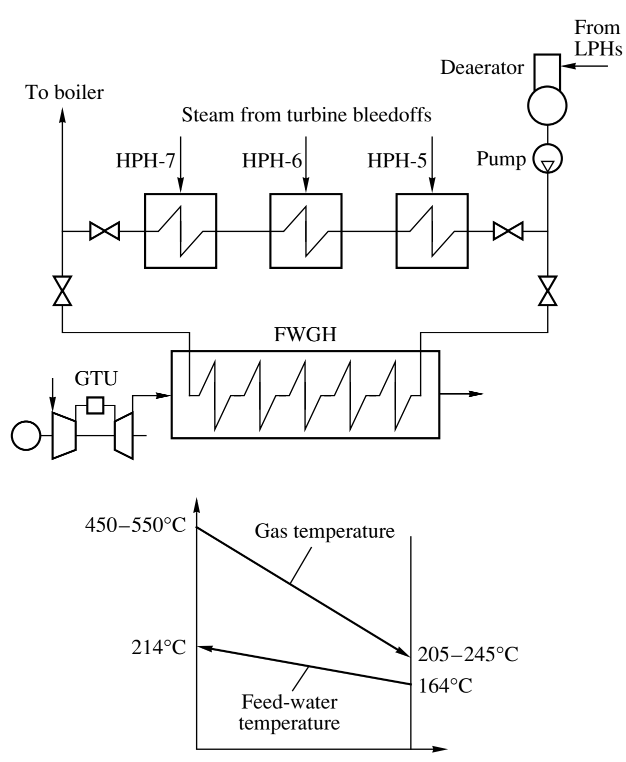
(iii) relative ease of installation of GTU and FWGH at a power plant in operation,

(iv) an increase in power mainly due to installation of a gas turbine and switching-off of high-pressure heaters (HPHs) of a steam turbine,

(v) improved cogeneration plant maneuverability, and

(vi) the possibility to the increase heat supply from a CCI with a cogeneration-type turbine while operating according to cogeneration schedule.

The use of CCI–Rs can appear to be especially profitable for power units with type T-110/120-12.8 turbines, which are installed at domestic cogeneration plants in amounts of more than 160 and most of which operate in regions with a deficit of electric power. Due to these reasons, a calculation study has been conducted to determine the advisability of substitution of feed water heating by GTU waste gases for high-pressure regeneration in a T-110-12.8 turbine, which operates in condensing and cogeneration modes.



Scheme of feed water heating by GTU waste gases. Changes in gas and feedwater temperatures in FWGH.

The scheme of such substitution is shown in the figure. The FWGH is arranged in parallel with the HPHs. Feedwater is heated in the FWGH to the same temperature as it is done in HPHs with the use of steam from STU bleedoffs. A comparison of the fuel saving due to changeover from a STU with a T-110/120-12.8 turbine to the CCI–R while operating in the condensing and cogeneration modes was conducted by comparing the absolute electric efficiencies of these installations,

$$\Delta\bar{\eta}_e = \frac{\eta_e^{\text{CCI-R}} - \eta_e^{\text{STU}}}{\eta_e^{\text{STU}}}, \tag{1}$$

where $\eta_e^{\text{CCI-R}}$ and η_e^{STU} are the electric efficiencies of CCI–R and STU, respectively.

Heat-design calculations conducted at UTZ showed that, in the condensing regime with a normally operating regeneration system, the power of the installation is $N_{e0}^{\text{STU}} = 110$ MW. The corresponding steam and heat flowrates are $D_{e0}^{\text{turb}} = 398$ t/h and $Q_0^{\text{turb}} = 286.26$ MW. Hence, the gross efficiency of the steam turbine installation is $\eta_{e0}^{\text{STU}} = N_{e0}^{\text{STU}}/Q_0^{\text{turb}} = 0.384$. The amount of heat delivered to the feedwater in the HPHs is $\Delta Q_0^{\text{HPH}} = 22.96$ MW.

Taking into account the existing limitations on the steam flowrate through the low-pressure cylinder with HPSs switched-off, the live-steam flowrate to the turbine should be reduced to $D^{\text{turb}} = 370.3$ t/h. In this case, the power and the heat flowrate to the turbine unit will fall to 105 and 266.34 MW, respectively. At this power, an amount of heat required for feedwater heating is

$$\Delta Q^{\text{HPH}} \approx \frac{D^{\text{turb}}}{D_0^{\text{turb}}}\Delta Q_0^{\text{HPH}} = 21.36\ \text{MW}.$$

Therefore, when an FWGH is substituted for HPHs, it is assumed that the CCI–R operates with a steam-turbine power of $N_{e0}^{\text{STU}} = 105$ MW, while $\Delta Q^{\text{HPH}} = 21.36$ MW are transferred to the feedwater in the FWGH. In subsequent calculations, the efficiency of a boiler η_b is taken to be the same under STU operation both with switched-on and -off HPHs.

For the CCI–R under consideration, the efficiency is determined by the following correlation:

$$\eta_e^{\text{CCI-R}} = \frac{N_e^{\text{STU}} + N_e^{\text{GTU}}}{Q^{\text{turb}}/\eta_b + N_e^{\text{GTU}}/\eta_e^{\text{GTU}}}, \tag{2}$$

where the numerator gives the CCI power, the denominator gives the heat input to it, and N_e^{GTU} and η_e^{GTU} are the electric power and the efficiency of the GTU.

The heat that it is necessary to input in FWGH to provide transfer of ΔQ^{HPH} to the feedwater is

$$\Delta Q^{\text{FWGH}} = \Delta Q^{\text{HPH}}/\eta^{\text{FWGH}},$$

where η^{FWGH} is the FWGH efficiency.

The equation of GTU heat balance is

$$N_e^{\text{GTU}}/\eta_e^{\text{GTU}} = N_e^{\text{GTU}} + Q^{\text{FWGH}}. \tag{3}$$

Then, the electric power of a GTU having η_e^{GTU}, which is necessary to provide feedwater heating by GTU waste gases, is as follows:

$$N_e^{\text{GTU}} = \frac{\eta_e^{\text{GTU}}}{1 - \eta_e^{\text{GTU}}}\frac{\Delta Q^{\text{HPH}}}{\eta^{\text{FWGH}}}. \tag{4}$$

FWGH efficiency is determined from the following correlation:

$$\eta^{\text{FWGH}} = \frac{\theta_{ex}^{\text{GTU}} - \theta_{ex}^{\text{FWGH}}}{\theta_{ex}^{\text{GTU}} - \theta_{\text{a.a}}}, \tag{5}$$

where θ_{ex}^{GTU} and $\theta_{ex}^{\text{FWGH}}$ are the gas temperatures at the exit from GTU and FWGH, respectively, and $\theta_{\text{a.a.}}$ is the ambient air temperature.

If we substitute (4) in (2), we will obtain the following final correlation for CCI–R efficiency:

$$\eta_e^{\text{CCI-R}} = \frac{N_e^{\text{STU}} + \Delta N}{Q^{\text{turb}}/\eta_b + \Delta Q}, \tag{6}$$

where

$$\Delta N = \frac{\eta_e^{\text{GTU}}}{1 - \eta_e^{\text{GTU}}}\frac{\Delta Q^{\text{HPH}}}{\eta^{\text{FWGH}}}; \quad \Delta Q = \frac{1}{1 - \eta_e^{\text{GTU}}}\frac{\Delta Q^{\text{HPH}}}{\eta^{\text{FWGH}}}.$$

We see from (6) that the dependence of the $\eta_e^{\text{CCI-R}}$ on FWGH efficiency (which, in turn, is determined by the GTU waste-gases and the inlet feed-water temperatures) and the boiler and GTU efficiencies is rather sophisticated. It is difficult to analyze the effect of these parameters on $\eta_e^{\text{CCI-R}}$ in a general form because of their mutual effect on each other. Therefore, to solve a particular problem, it is necessary to choose a suitable GTU from the existing set of engines and then to assess all the indexes of the CCI formed. For this purpose, using the FWGH heat balance equation, the flowrate of gases through the FWGH (and GTU) is calculated,

$$G_g = \frac{\Delta Q^{\text{HPH}}}{c_{pg}\Delta\theta_g}, \tag{7}$$

where c_{pg} is the specific heat of the gases entering the FWGH (in all of the following estimates, it is assumed that $c_{pg} = 1.15$ kJ/(kg K)); $\Delta\theta_g = \theta_{ex}^{\text{GTU}} - \theta_{ex}^{\text{FWGH}}$ is a decrease in the GTU waste-gas temperature in the FWGH.

Table 1. Results of calculations of the possibility of using different GTUs with waste-gas flowrates of 60–80 kg/s for realization of CCI–Rs with a type T-110-12.8 turbine

Index	GTU (manufacturer)							
	GTU 10/95 (NPP Motor, Ufa)	GTD-16 (Mash-proekt, Nikolaev)	AL-31 (AO Saturn–Lyul'ka)	GTU-25PE (AO Avi-advigatel', Perm)	GT10B (Alstom)	LM2500 (General Electric)	NK-37 (SNTK Dvi-gateli NK, Samara)	GTG-25 (Mash-proekt, Nikolaev)
GTU gas flowrate, kg/s	62.4	72.0	61.0	78.4	80.4	68.9	102.0	87.5
GTU electric power, MW	10.0	17.5	20.0	25.0	24.8	22.45	25.0	27.5
GTU electric efficiency	0.31	0.35	0.365	0.39	0.342	0.357	0.364	0.36
GTU waste-gas temperature, °C	478	432	520	451	543	525	428	485
FWGH efficiency	0.600	0.556	0.634	0.576	0.650	0.637	0.552	0.606
GTU gas flowrate necessary for feedwater heating in FWGH, kg/s	69.22	82.95	60.14	76.67	56.10	59.21	84.4	67.52
Amount of heat, MW:								
input in GTU combustion chamber	32.26	50	54.79	64.10	72.51	62.89	68.68	76.39
input to FWGH	22.26	32.5	34.79	39.1	47.71	40.44	43.68	48.89
transferred to feedwater	13.36	18.07	22.06	22.52	31.01	25.76	24.11	29.63

The temperature difference at the exit from the FWGH is

$$\Delta t_{\mathrm{FWGH}} = \theta_{\mathrm{ex}}^{\mathrm{FWGH}} - t_{\mathrm{p}},$$

where $t_{\mathrm{p}} = 164°\mathrm{C}$ is the water temperature downstream of the feed-water pump.

We have that $\Delta t_{\mathrm{FWGH}} \approx 36°\mathrm{C}$; then, $\theta_{\mathrm{ex}}^{\mathrm{FWGH}} = 200°\mathrm{C}$. GTU waste-gas temperature $\theta_{\mathrm{ex}}^{\mathrm{GTU}}$ changes within the range of 450–550°C; therefore, from formula (7) it follows that we must have a GTU with a gas flowrate of 60–80 kg/s to realize the CCI–R under consideration. The list of GTUs chosen in accordance with the above requirement is presented in Table 1 [1]. In addition, the values of the parameters that enable us to assess the suitability of these GTUs for use in the CCI under consideration are presented in Table 1. With the known $\theta_{\mathrm{ex}}^{\mathrm{GTU}}$ and the assumed $\theta_{\mathrm{ex}}^{\mathrm{FWGH}}$, we can calculate from (7) the necessary flowrate of the heating gases through the FWGH. The comparison of this necessary flowrate with the actual one shows that GTU-10/95 and GTD-16 turbines are not suitable for using in the CCI–R under consideration because of a low gas flowrate, while the GT10B turbine is not suitable due to the high gas flowrate and high temperature of the waste gases. AL-31ST and GTU-25PE turbines and, to a certain extent, the NK-37 turbine meet the given conditions. (For the last, the high gas flowrate is balanced by the low gas temperature.) It is necessary to emphasize that, for AL-31ST and GTU-25PE turbines, the calculated amount of heat transferred to the feedwater is essentially equal to the necessary $\Delta Q^{\mathrm{HPH}} = 21.36$ MW.

Table 2 shows the results of detailed calculations of the economic indexes of the CCI–R and the temperature differences in the FWGH for the gas turbines chosen. The total amount of heat delivered to the steam turbine and the combustion chamber of the GTU is determined from the expression

$$Q_{\mathrm{SSI\text{-}R}} = Q^{\mathrm{turb}}/\eta_{\mathrm{b}} + N_{\mathrm{e}}^{\mathrm{GTU}}/\eta_{\mathrm{e}}^{\mathrm{GTU}}$$

and its electric efficiency from Eq. (2).

It is seen from the analysis of the data in Table 2 that the fuel savings calculated by correlation (1) are maximum for a CCI with a GTU-25PE turbine and reaches almost 4%.

The temperature differences at the exit from the FWGH, which are determined by means of formula

$$\delta t_{\mathrm{ex}}^{\mathrm{FWGH}} = \theta_{\mathrm{ex}}^{\mathrm{GTU}} - \frac{\Delta Q^{\mathrm{HPH}}}{c_{pg} G_{\mathrm{g}}} - t_{\mathrm{p}}$$

are essentially the same for both GTU-25PE and AL-31ST turbines and provide necessary heating of feedwater. This is even more the case for the NK-37 gas turbine.

Thus, the implementation of GTU-25PE, AL-31ST, and NK-37 turbines with corresponding FWGH, which uses GTU waste-gas heat, makes it possible to obtain a CCI–R of 125- to 130-MW capacity with an electric efficiency of 36.5–37.1%, that is, to provide a fuel savings of 2.5–4.0% as compared to the original steam-turbine unit with a T-110-12.8 turbine. The final choice of type of GTU and the conclusion on the advisability of implementation of the modernization under consideration should be done with an account of payback time,

Table 2. Results of calculations of fuel savings while substituting of GTU waste gases for HP bleedoffs of a T-110-12.8 steam turbine

Index	GTU		
	AL-31ST	GTU-25PE	NK-37
Condensing mode of operation			
CCI electric power, MW	125.0	130.0	130
Amount of heat delivered to CCI, MW	341.18	350.49	355.07
Electric efficiency of CCI–R, %	36.6	37.1	36.6
Electric efficiency of STU, %	35.7	35.7	35.7
Increase in efficiency of CCI–R as compared to STU, %	0.9	1.4	0.9
Fuel savings in CCI–R as compared to STU, %	2.52	3.92	2.52
Gas temperature downstream of FWGH, °C	204.5	205.6	239.3
Temperature difference at FWGH outlet, °C	40.5	41.6	75.3
Cogeneration mode of operation			
Electric power of CCI–R, MW	–	130	–
Heat power of cogeneration bleedoffs, MW:			
STU	–	203.7	–
CCI–R	–	203.7	–
Coefficient of useful fuel utilization:			
CCI–R	–	0.896	–
STU	–	0.881	–
Fuel savings in CCI–R as compared to STU, %	–	1.7	–

as well as many other factors. Such an analysis is beyond the scope of our paper.

Below, we give a comparison of the economic indexes of the CCI–R with GTU-25PE and STU with a T-110/120 cogeneration turbine under conditions of their operation in the rated (cogeneration) regime. Because we are comparing installations that are already set up, the comparison is carried out on the basis of the fuel heat utilization coefficient $K_{\rm use}$ (naturally, a similar comparison can be made on the basis of the efficiency or consumption of a conditional fuel for producing electricity and heat).

The relative savings of fuel in a CCI–R under the cogeneration mode of operation as compared to an STU is

$$\Delta\bar{K}_{\rm use} = \frac{K_{\rm use}^{\rm CCI\text{–}R} - K_{\rm use}^{\rm STU}}{K_{\rm use}^{\rm STU}}. \tag{8}$$

At the rated cogeneration regime, the steam-turbine installation has an electric power of 110 MW; the steam and heat flowrates to the turbine are $D_{\rm e.c0}^{\rm turb}$ = 480 t/h and $Q_{\rm h0}^{\rm turb}$ = 330.89 MW, respectively. The heat load of the cogeneration bleedoffs $Q_{\rm h0}$ = 203.7 MW (175 Gcal/h) (with two-staged heating of the heating-network water, the pressure in the upper cogeneration bleedoff is 0.1 MPa and the temperature of the return water is 50.8°C). The gross $K_{\rm use}^{\rm STU}$ coefficient is determined as follows:

$$K_{\rm use}^{\rm STU} = \frac{N_{\rm e.c0}^{\rm STU} + Q_{\rm h0}}{Q_{\rm 0h}^{\rm turb}/\eta_{\rm b}} = 0.881.$$

When HPHs are switched off, the steam flowrate to the heating-network heaters increases by 64.2 t/h. However, due to the inadmissibility of increasing bending strengths in the working blades of the last intermediate-pressure cylinder (IPC) stage, we must reduce the live steam flowrate by approximately 13.4%, i.e., down to $D_{\rm e.c}^{\rm turb}$ = 415.8 t/h. The power of the high-pressure cylinder will be reduced to approximately the same extent, the power of the section between the first and the second bleedoffs will be reduced to a considerably lesser extent, and the power of the section between the second and the third bleedoffs will be reduced to an even smaller extent. In the regime under consideration, the other stages will generate the same power as when the steam turbine is operating at the rated flowrate of the live steam. On the whole, we can suppose that the steam turbine power will be reduced and constitute $N_{\rm e.c}^{\rm STU}$ = 105 MW. When this happens, the heat capacity of the bleedoffs will not change, i.e., $Q_{\rm h} = Q_{\rm h0}$ = 203.7 MW. With a decrease in the live steam flowrate, the amount

of heat delivered to the turbine decreases also; therefore, $Q_{\mathrm{h}}^{\mathrm{turb}}$ will equal 286.64 MW.

For the CCI–R under consideration, the gross K_{use} will be

$$K_{\mathrm{use}}^{\mathrm{CCI-R}} = \frac{N_{\mathrm{e.c}}^{\mathrm{STU}} + N_{\mathrm{e}}^{\mathrm{STU}} + Q_{\mathrm{h}}}{Q_{\mathrm{h}}^{\mathrm{turb}}/\eta_{\mathrm{b}} + N_{\mathrm{e}}^{\mathrm{GTU}}/\eta_{\mathrm{e}}^{\mathrm{GTU}}} = 0.896. \qquad (9)$$

In reality, the actual values of K_{use} will be somewhat lower both for the STU and CCI–R, because of heat losses due to the small discharge of cooling steam to the condenser. This, however, does not change the final estimates. It follows from (8) that fuel savings in cogeneration mode of operation due to substitution of the CCI–R for the STU will be 1.7%. When this happens, the heat load of the installation remains the same and the electric power increases from 110 to 130 MW.

For the CCI–R under consideration, the FWGH—or more exactly the water-heating heat-recovery boiler—constitutes a new piece of equipment, which is manufactured on the basis of an individual design. In the catalog of the Taganrog Krasnyi Kotel'shchik Boiler Works, a horizontal heat-recovery boiler of 37-MW (32-Gcal/h) capacity is listed, which heats condensate by waste gases of a GT-10 gas turbine manufactured by the Joint Enterprise SP ABB–Nevskii. The capacity of this turbine is 24.63 MW, its efficiency is 34.2%, the temperature of the waste gases is 534°C, and the gas flowrate is 79 kg/s [2]. The heat-recovery boiler is capable of heating 330 t/h of condensate from 79 to 175°C. The temperature of the stack gases downstream of the boiler is 110°C. The heat-transfer surfaces of such a boiler are made of spirally finned tubes arranged horizontally and joined by vertical manifolds. Therefore, the construction of such a boiler brings no specific problems.

Thus, topping of the turbine units at cogeneration plants with UTZ T-110/120-12.8 steam turbines by means of GTUs can be accomplished using the scheme with substitution of the waste gases of GTU for the high-pressure heaters of the turbine regeneration system for heating feed water. To realize this scheme, it is necessary to use GTUs of 25-MW capacity with a waste-gas flowrate of 60–80 kg/s. From the set of domestic GTUs, the GTU-25PE, AL-31ST, and NK-37 seem to be the most suitable for the above objective. Such topping makes it possible to increase electric power to 130 MW, while operating in condensing and cogeneration modes, and reduce fuel consumption by 2.5–4.0% in condensing regime and by 1.5–2.0% in cogeneration regime as compared to the STU in operation.

There exists a possibility for further improvement of the scheme with a T-110/120-12.8 steam turbine considered. For example, this can be done by additional utilization of the heat of stack gases of the water-heating heat-recovery boiler for partial heating of the heating-network water.

In conclusion, we should emphasize that the economic advisability of implementation of CCI–Rs should be determined by technical and economical calculations. We hope that such a scheme will be used while replacing T-110/120-12.8 turbines by new similar ones with simultaneous addition of GTUs to realize the combined-cycle scheme with substitution for high-pressure regeneration.

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